Hybrid dynamic modeling of shearer’s drum driving system and the influence of housing topological optimization on the dynamic characteristics of gears

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Abstract

The drum shearer is one of the main equipments of the long-wall mining system that has been widely used in coal mining for decades. Influenced by large fluctuation and intensive impact of drum load, the drum driving system is a weak part of the drum shearer due to the large dynamic deformation on ranging arm housing and the changes of gear meshing state. Since previous studies did not consider the coupling effects of housing and gearing transmission system, this study is concerned with hybrid finite element/lumped parameter dynamic modeling of the housing-transmission coupled system. In order to model the drum driving system, it is subdivided into two substructures: housing and transmission system. The housing is modeled by finite element method while the transmission system is modeled by lumped parameter method. The dynamic sub-structuring method (DSM) is used to develop a hybrid dynamic model, taking elastic coupling between the housing and transmission system into consideration. Then the influence of housing topological optimization on the dynamic characteristics of drum driving system is also analyzed through the hybrid dynamic model. The housing topological optimization is conducted through the Optistruct solver in Hypermesh, aiming at increasing the natural frequency of housing. It can be concluded that the ranging arm housing topological optimization could reduce the dynamic deformation of the housing and thus also reduce the equivalent mesh misalignment. However, the influence of housing topological optimization on the dynamic meshing force is not very significant.

Keywords: Drum shearer, Gear transmission system, Craig method, Dynamic sub-structuring method, Topological optimization

1. Introduction

Long-wall mining has been the dominant global coal mining method for decades and the drum shearer is one of the main equipments of the long-wall mining system (Kingshott and Graham, 1998). The appearance of a drum shearer is shown in Fig. 1. The gear failure accounts for more than 34% of the shear failure (Zhou, 2011), and the drum driving system is one of the weakest parts of the drum shearer (Liu et al., 2015). It composed by an asynchronous motor, a multistage gearing system, a drum and a ranging arm housing, which is shown in Fig. 2. As the ranging arm housing is cantilevered, and the drum load is generally large and accompanied by high impact, the large housing deformation will result in unbalance loading on the gear tooth. Therefore, in order to design a reliable and cost effective drum driving system, it is necessary to study the coupling effect of the housing and internal transmission system. It is also valuable to investigate the effect of housing topological optimization on the dynamic characteristics of drum driving system.

As a multistage geared electromechanical coupling system, the systematic dynamic responses of the drum driving system have drawn extensive attention from the research community. Dolipski et al. (2000a, 2001b) constructed a dynamic model of the drum driving system of a KSW-500 shearer using the lumped parameter gear method and analyzed the effects of the hauling speed and coal/rock hardness on the dynamic meshing force. By considering the
time-varying mesh stiffness of the gears, Jiang et al. (2016) investigated the nonlinear vibration characteristics of the drum driving system using a Poincaré map and highlighted the practicality of employing chaos characteristics as a means of crack detection. By taking into account the dynamic characteristics of the asynchronous motor, Liu et al. (2015) constructed an electromechanical coupling model of the drum driving system that included the speed-change process, and used this model to study the electromechanical coupling properties of the system under impact load. Shu et al. (2015) and Li et al. (2016) examined the gear load sharing behavior of a novel cutting transmission system of the drum shearer. However, the effects of the deformation of the ranging arm housing on the transmission system were neglected in the above-mentioned studies.

Gearbox housing is one of the most critical components of a gearing transmission system. The effects of stiffener layout on the vibration and noise of gearbox housing was studied (Inoue et al., 2002), and the result showed that the stiffener layout for low noise is identical to the layout for low vibration, except in the case of one particular vibration mode for the simple flat plate. But the influence of the interior gearbox air-cavity on its vibratory dynamic response was ignored. By considering the acoustic-structural interaction mechanisms, Abbes et al. (2008) used a modal analysis method to evaluate the elasto-acoustic modal characteristics of a gearbox, and the effect of the fluid inside the gearbox on the vibration response is discussed. The process of spreading disturbance power through the elastic structure, especially the role of the gearbox housing, was explained by Ognjanović and Kostić (2012). By finding the mode order and panel with maximum acoustic contributions, Wang et al. (2014) searched to reduce the radiated noise of the gearbox. However, the effects of gearbox housing on the equivalent mesh misalignment and dynamic meshing force were not studied. Additionally, the coupling effect of housing and internal transmission system was neglected in the above-mentioned studies.

In this study, a hybrid dynamic model is constructed including an asynchronous motor, the parallel-axis gears, the shafts, the bearings, the planetary gears, the drum, and housing. The housing is represented by a finite element grid, and the classic reduction process of Craig-Bampton method is employed to get a reduced model of condensed stiffness and mass matrices. The shafts are represented by Timoshenko beam elements taking the bend, rotation and shear deformation into consideration. Other components are modeled by lumped parameter method. At last, two different housing (original and optimal) are compared under the same load case. The numerical results show that the housing topological optimization could effectively reduce the gears’ equivalent mesh misalignment.

![Fig. 1 The appearance of a long-wall shearer. The drum driving system is heightened with the red rectangle.](image1)

![Fig. 2 The drum driving system. The cutting power transfers from the motor to the drum, through two-stage parallel-axis gears and two-stage planetary gears.](image2)

2. Dynamic modeling of drum driving system

2.1 Modeling ranging arm housing

Figure 3 illustrates the finite element model of the ranging arm housing, and it can be seamlessly integrated with other components through the mass and stiffness matrices. A grid-independent examination was conducted when constructing the finite element model. The model is composed of 64,789 three-directional tetrahedron elements and has 52,554 degrees of freedom (DOFs). First, the mass and stiffness matrices of the ranging arm housing are extracted from the ABAQUS software, and are referred to as the primitive matrices. Then, the orders of the primitive matrices are reduced using the Craig-Bampton transformation (Craig and Bampton, 1968) in order to improve the dynamic computation efficiency and reduce the storage space required for the simulation results.
\[
\begin{bmatrix}
\mathbf{u}^h \\
\mathbf{u}^b
\end{bmatrix} = 
\begin{bmatrix}
\mathbf{\Phi} & \mathbf{0} \\
\mathbf{0} & \mathbf{I}
\end{bmatrix} 
\begin{bmatrix}
\mathbf{q}^h \\
\mathbf{q}^b
\end{bmatrix} = 
\mathbf{T}_c \{q^*\}
\]

where \(\mathbf{u}^h\) represents the generalized displacements of the internal points; \(\mathbf{u}^b\) represents the generalized displacements of the boundary points; \(\mathbf{\Phi}\) is the truncated modal matrix of the fixed-interface ranging arm housing; \(\mathbf{\Phi}\) is the constraint modal matrix, with each column representing the values assumed by the DOFs at the internal points for unit value of the degrees-of-freedom at an boundary point; \(\mathbf{I}\) is the identity matrix; \(\mathbf{q}^h\) is the mode coordinate; and \(\mathbf{T}_c\) is the Craig–Brampton transformation matrix.

When calculating \(\mathbf{T}_c\), the inner surface of the bearing bore is selected as the coupling interface, and the bearing bore is assumed to be undeformable. Accordingly, the center of the bearing bore is set as the boundary point, and rigid elements are used to connect the boundary point and the points on the coupling interface, as illustrated in Fig. 3(b). From the motor end to the drum end, as well as from the positive direction to the negative direction of the z axis, the boundary points are consecutively numbered as BP1 to BP17.

\[
\mathbf{M}^h = \mathbf{T}_c^T \mathbf{[M]} \mathbf{T}_c
\]

\[
\mathbf{K}^h = \mathbf{T}_c^T \mathbf{[K]} \mathbf{T}_c
\]

where \(\mathbf{M}\) and \(\mathbf{K}\) are the primitive mass and stiffness matrices extracted from the ABAQUS software, both of which are square matrices on the order of 52,554; \(\mathbf{M}^h\) and \(\mathbf{K}^h\) are the condensed matrices. Because the truncated mode and constraint mode have orders of 60 and 102, respectively, the condensed matrices are square matrices with an order of 162.

### 2.2 Modeling gear shafts

The gear shaft is a cylindrical structure with a varying cross section. The Timoshenko beam element can effectively consider the bending, torsion, and shear deformation of a gear shaft simultaneously. Fig. 4(a) illustrates a two-node Timoshenko beam element, with each node possessing 6 DOFs. A gear shaft is divided into multiple Timoshenko beam elements using a few shaft nodes, which are set at the shaft diameter variation point, bearing support middle point, gear installation middle point, and power input point. The mass and stiffness matrices of any element for gear shaft \(s_i\) are denoted as \(\mathbf{M}_{i,j+1}\) and \(\mathbf{K}_{i,j+1}\), respectively, where \(j = 1, 2, \ldots, n\), and \(n\) is the number of elements of gear shaft \(s_i\).
Stringer (2008) provided details of the Timoshenko beam element matrices; therefore, these will not be discussed in this paper. Figure 4(b) is an example of a five-node and four-element gear shaft, whose mass matrix $M_j^{oo}$ and stiffness matrix $K_j^{oo}$ are obtained by superimposing the corresponding element matrices in the proper order shown in Fig. 4(c).

2.3 Modeling parallel-axis gears

Parallel-axis gear is regarded as rigid disk with concentrated mass and moment of inertia. The generalized displacements of gear $g_j$ are defined as $u^g = [x^g, y^g, z^g, \theta_x^g, \theta_y^g, \theta_z^g]^T$. The dynamic equation of a pair of parallel-axis gears can be obtained using the lumped parameter method:

$$
\begin{bmatrix}
M_1^{e1} & M_2^{e1} \\
M_1^{e2} & M_2^{e2}
\end{bmatrix}
\begin{bmatrix}
\ddot{u}_1^{e1} \\
\ddot{u}_2^{e2}
\end{bmatrix} = 
\begin{bmatrix}
R_{n,1} \\
R_{n,2}
\end{bmatrix}
$$

where $M_j^{oo} = \text{diag}(m_j^{oo}, m_j^{oo}, m_j^{oo}, I_j^{oo}, I_j^{oo}, I_j^{oo})$, $j = 1, 2$, where “diag” denotes a diagonal matrix; $m_j^{oo}$ and $I_j^{oo}(k = x, y, z)$ represent the mass of gear $g_j$ and moment of inertia about the $k$ axis at the center of mass, respectively; $R_{n,j} = [F_j, T_j]^T$, $j = 1, 2$ is the engaging load of gear $g_j$ (both engaging force $F_j$ and engaging moment $T_j$ are 3D vectors); $R_{n,j}$ is a nonlinear function of $u_1^{e1}$ and $u_2^{e2}$.

To obtain the engaging force and engaging moment, the load distribution along the direction of tooth length is calculated using the pseudo-3D thin-slice approach (Velex and Maatar, 1996). As illustrated in Fig. 5, this approach considers the gear as having being constructed by stacking thin slices of identical thickness along the direction of tooth width and assumes that the linear-elastic contact of the driving gear and driven gear occurs at a series of micro engaging elements $dM$. Thus, the engaging force and engaging moment of gear $g_j$ are expressed as follows:

$$
F_j = \int_b dF_j = \left( \int_b k_1 \delta(M') \right) \cdot (\mathbf{n}_j^{oo})
$$

$$
T_j = \int_b dT_j = \left( \int_b k_2 \delta(M') \right) \cdot \text{cross}(\mathbf{n}_j^{oo}, \mathbf{n}_j^{oo})
$$

where $b$ is the tooth width; $dF_j$ is the vector representing the dynamic meshing force at the micro engaging element
$dM;\;dT_j$ is the engaging moment tensor of $dF_j$ with respect to the center of mass of gear $g_j$; $k_r$ is the mesh stiffness of each engaging element; $n^{\sigma}$ is the normal vector of the mesh plane of gear $g_j$; $r^{\sigma}_j$ is the vector pointing to the engaging point $M'$ from the center of mass of gear $g_j$; $\text{cross}(a, b)$ means the cross product of two vectors $a$ and $b$, the calculation result is a vector; and $\delta(M')$ is the scalar of normal deviation at the middle point $M'$ of engaging element $dM$.

To compute the load on a micro engaging element, it is necessary to measure the equivalent normal deviation of the element. When the engaging elements are of a sufficiently fine scale, the normal deviation at the middle point $M'$ of a particular element can be considered to represent the normal deformation of the entire element.

$$\delta(M') = \sum_{j=1}^{2} \dot{\text{dot}} \left( R^{\sigma}_{gj} + \text{cross} \left( r^{\sigma}_j, \Omega^{\sigma}_{gj} \right), n^{\sigma} \right) - e^i$$

where $R^{\sigma}_{gj} = \begin{bmatrix} x^{\sigma}_{gj}, y^{\sigma}_{gj}, z^{\sigma}_{gj} \end{bmatrix}^T$ is the vector of rigid-body displacement for the gear $g_j$; $\Omega^{\sigma}_{gj} = \begin{bmatrix} \theta^{\sigma}_{xgj}, \theta^{\sigma}_{ygj}, \theta^{\sigma}_{zgj} \end{bmatrix}^T$ is the vector of 3D rotational angle of the gear; $e^i$ is the scalar of tooth surface error at point $M'$; $\dot{\text{dot}}(a, b)$ means the dot product of two vectors $a$ and $b$, the calculation result is a scalar.

### 2.4 Modeling planetary gears

Abousleiman et al. (2007) constructed a planetary gear dynamic model of which both ring gear and planet carrier are flexible. Although this model has reasonable theoretical accuracy, the modeling process is complex. Because the planetary gear system was not the focus of the present study, we adopted the lumped parameter method to construct a pure torsional planetary gear dynamic model (Liu and Qin, 2015). Figure 6 illustrates the sketch of pure torsional model of a planetary gear, in which $\theta^s_r, \theta^p_r, \theta^o_r$, and $\theta^s_r (n = 1, 2, \ldots, N)$ denote the angular displacements of the sun gear, planet carrier, ring gear, and planet gear on the $x$-$y$ plane, respectively. The differential equations of the planetary gear set are expressed as follows:

$$\begin{cases}
I^r \ddot{\theta}_r^s + \sum_{n=1}^{N} (k_m \delta_m) r_n^s + k_r \theta_r^s = 0 \\
I^r + Nm^r (r^p)^2 \ddot{\theta}_r^p - \sum_{n=1}^{N} (k_m \delta_m \cos \alpha_r + k_m \delta_m \cos \alpha_r \cos \alpha_r) r^p - k_r \theta_r^p = 0 \\
I^r \ddot{\theta}_r^p + \sum_{n=1}^{N} (k_m \delta_m) r_n^p + k_r \theta_r^p = 0 \\
I^r \ddot{\theta}_r^o + (k_m \delta_m - k_m \delta_m) r_n^o = 0 \quad (n = 1, 2, \ldots, N)
\end{cases}$$

Fig. 5 Parallel-axis gear model. The load distribution along the contact line is calculated using the pseudo-3D thin-slice approach, which assumes that the gear is divided into a number of slices (the width of which is $dM$).
where $\alpha_i$ and $\alpha_s$ are the inner and outer engaging angles, respectively; $I^i (i = s, c, r, 1, 2, ..., N)$ is the moment of inertia of component $i$ about its geometric center; $m^r$ is the mass of the planet gear; $r_i^r (i = x, r, 1, 2, ..., N)$ is the base circle radius of component $i$; $r^c$ is the radius of the planet carrier; $\delta_i^m$ and $\delta_s^m$ are the normal deviations on the inner and outer engaging lines, respectively.

$$[M^p] \{\ddot{u}^p\} + [K^p] \{u^p\} = \{0\}$$

(9)

where $u^p = [u^s, u^c, u^r, u^1, u^2, ..., u^N]^T$ is the generalized displacement of the planetary gear system; $u^i = [0, 0, 0, 0, 0, \theta_i^x]^T (i = s, c, r, 1, 2, ..., N)$; $M^r = \text{diag}(M^s, M^c, M^r, M^1, M^2, ..., M^N)$ is the global mass matrix of the planetary gear set; $M' = \text{diag}(0, 0, 0, 0, 0, I^i) (i = s, c, r, 1, 2, ..., N)$; and $K^p$ is the global stiffness matrix of the planetary gear set.

2.5 Modeling electric motor

The asynchronous motor without control is the power source of the drum driving system. When the load increases, the rotating speed of the motor will decrease, vice versa. That means the rotating speed is the function of the load. The mechanical characteristic of the electric motor reflects the relationship between the torque and the rotating speed. The torque-speed function is derived as Eq. 10, the Kloss’s Law (Gerling, 2015), which has an advantage in obtaining the parameters.

$$T = \frac{2T_{\text{max}}}{s/s_m + s_m/s}$$

(10)

where $T_{\text{max}}$ is the maximum torque; $s_m$ is the critical slip, $s_m = 0.0746$ here; $s$ is the slip, and the rotating speed is...
derived as \( n = (1 - s) n_s \), here \( n_s \) is the synchronous speed.

### 2.6 Coupling model of drum driving system

The drum driving system consists of the ranging arm housing, asynchronous motor, parallel-axis gears, gear shafts, planetary gears, and the drum. The deformation of the ranging arm housing is represented by the boundary points on the housing. These components are connected through various means such as fixed connections, bearing connections, coupling connections, and meshing connections to form the complete coupling model of the drum driving system, as illustrated in Fig. 7. The generalized displacements of the complete system are

\[
\mathbf{u} = \begin{bmatrix} u_h^1, u_g^e, u_p^r, u_s^n, u_m^d, u_d^e \end{bmatrix}^T
\]

where the superscripts denote different components, i.e., ranging arm housing \((h)\), parallel-axis gears \((g)\), planetary gears \((p)\), gear shafts \((s)\), motor \((m)\), and drum \((d)\); and

\[
\mathbf{u}' = \begin{bmatrix} u_1^1, u_2^2, \ldots, u_N^N \end{bmatrix}^T \quad (i = g, p, s, m, d)
\]

Because the generalized displacements of each component \( \mathbf{u}' \) are defined under the unified coordinates, no coordinate transformation is necessary. Hence, the coupling dynamic equation of the drum driving system can be constructed in the following two steps: 1) directly construct the primitive equation using the coefficient matrix of each component, and 2) obtain the final dynamic equation using matrix transformation. The primitive equation is expressed as follows:

\[
[m][\ddot{\mathbf{u}}] + [k][\mathbf{u}] = \{r\}
\]

where \( m = \text{diag}(M_h^h, M_g^g, M_p^p, M_s^s, M_m^m, M_d^d) \), \( k = \text{diag}(K_h^h, K_g^g, K_p^p, 0, 0) \), and \( r = [0, 0, 0, 0, R_m^m, R_d^d]^T \); specifically, \( M_m^m = \text{diag}(m_h^h, m_g^g, m_p^p, I_m^m, I_m^m, I_m^m) \) is the mass matrix of the motor rotor, \( M_d^d = \text{diag}(m_g^g, m_s^s, I_s^s, I_s^s, I_s^s) \) is the mass matrix of the drum, \( R_m^m = [0, 0, 0, 0, T_m^m]^T \) is the asynchronous motor driving load, \( R_d^d = [F_{d_h}^d, F_{d_g}^d, F_{d_p}^d, 0, 0, T_d^d]^T \) is the drum load.

Fig. 7 Coupling model of drum driving system. The housing deformation is coupled with the transmission system through the boundary points. All components are connected using four kinds of connections such as: fixed connection, bearing connection, coupling connection, and meshing connection.

Until now, the primitive equation has not considered the four types of connections illustrated in Fig. 7. Considering the connection between any two nodes \( i \) and \( k \) as an example, the transformation of the corresponding node coefficient matrix can be illustrated as follows:
where \( \mathbf{u}^i \) and \( \mathbf{u}^k \) are the generalized displacements of nodes \( i \) and \( k \), respectively, both of which are 6D vectors. Hence \( \mathbf{M}^i, \mathbf{M}^k, \mathbf{K}^i \), and \( \mathbf{K}^k \) are square matrices with an order of six; and \( \mathbf{R}^i \) and \( \mathbf{R}^k \) are 6D vectors denoting the external load.

1) Matrix transformation of fixed connection

Since two fixed-connected nodes have identical displacements, speeds, and accelerations, they can be combined in a single node. The transformation for Eq. (13) is as follows:

\[
\begin{bmatrix}
\mathbf{M}^i + \mathbf{M}^k
\end{bmatrix}
\begin{bmatrix}
\mathbf{u}^i \\
\mathbf{u}^k
\end{bmatrix}
+ 
\begin{bmatrix}
\mathbf{K}^i + \mathbf{K}^k
\end{bmatrix}
\begin{bmatrix}
\mathbf{u}^i \\
\mathbf{u}^k
\end{bmatrix}
= 
\begin{bmatrix}
\mathbf{R}^i + \mathbf{R}^k
\end{bmatrix}
\] (14)

where \( \mathbf{u}^k \) vanishes, and the corresponding rows and columns in the coefficient matrix are removed.

2) Matrix transformation of bearing and coupling connections

Both bearing and coupling connections are elastic connection in nature; therefore, their matrix transformations have similar forms:

\[
\begin{bmatrix}
\mathbf{M}^i \\
\mathbf{M}^k
\end{bmatrix}
\begin{bmatrix}
\mathbf{u}^i \\
\mathbf{u}^k
\end{bmatrix}
+ 
\begin{bmatrix}
\mathbf{K}^i + \mathbf{K}^c \\
-\mathbf{K}^c
\end{bmatrix}
\begin{bmatrix}
\mathbf{u}^i \\
\mathbf{u}^k
\end{bmatrix}
= 
\begin{bmatrix}
\mathbf{R}^i \\
\mathbf{R}^k
\end{bmatrix}
\] (15)

where \( \mathbf{K}^c = \text{diag}(k_c, k_c, k_c, k_{x_m}, k_{y_m}, 0) \) when the bearing connection is simulated, and \( \mathbf{K}^c = \text{diag}(0, 0, 0, 0, 0, k_m) \) when the coupling connection is simulated.

3) Matrix transformation of meshing connection

When two nodes are meshing connected, their mass and stiffness matrices are not altered. Instead, coupling is achieved via the engaging force and moment on the right side of Eq. (13). Section 2.3 has described the calculation method of the meshing force and moment.

\[
\begin{bmatrix}
\mathbf{M}^i \\
\mathbf{M}^k
\end{bmatrix}
\begin{bmatrix}
\mathbf{u}^i \\
\mathbf{u}^k
\end{bmatrix}
+ 
\begin{bmatrix}
\mathbf{K}^i \\
-\mathbf{K}^i
\end{bmatrix}
\begin{bmatrix}
\mathbf{u}^i \\
\mathbf{u}^k
\end{bmatrix}
= 
\begin{bmatrix}
\mathbf{R}^i + \mathbf{R}_{m,i} \\
\mathbf{R}^k + \mathbf{R}_{m,k}
\end{bmatrix}
\] (16)

where \( \mathbf{R}_{m,i} \) and \( \mathbf{R}_{m,k} \) are both nonlinear functions of \( \mathbf{u}^i \) and \( \mathbf{u}^k \), demonstrating the nonlinear connection.
between gear nodes \( i \) and \( k \).

Considering all the connections illustrated in Fig. 7, the complete model of the drum driving system is finally obtained as follows:

\[
[M]\{\ddot{U}\} + [C]\{\dot{U}\} + [K]\{U\} = \{R\}
\]  

(17)

where \( M \), \( K \), and \( R \) are respectively obtained from \( m_k \), \( k \), and \( r \) in Eq. (12) via Eqs. (14-16), and the damping matrix is constructed using proportional damping, i.e., \( C = \alpha M + \beta K \), where \( \alpha \) and \( \beta \) are the proportional damping coefficients.

3. Validation of the hybrid dynamic model

Before extensive studies are performed, validation of the hybrid dynamic model is conducted. It should be noted that the rated power of drum driving system for simulation is 1,200 kW, while the rated power of drum driving system for experiment is 500 kW. This is because of the lack of geometry parameters for experimental drum driving system. Although the experimental drum driving system is smaller than the simulation one, the structures of them are similar. It is believed that some similar phenomenon can be found from both experimental result and simulation result. Two load cases are used for the validation of the hybrid dynamic model in this section.

a) Stationary random load, for which the mean value of cutting moment is 515.1 kN·m.

b) Load increasing process, for which the mean value of cutting moment increased from 0 to 515.1 kN·m within 18 seconds.

Figure 8 shows the experimental system, in which the acceleration sensors are set on the range arm housing, and the LMS data acquisition system is placed on the shearer body. A wooden box and a steel protective cover are used to protect the vibration test system from crushed coal. The experimental cutting process lasts more than 10 minutes, and it contains both two load cases: a) stationary random load, and b) load increasing process. And then the validation of the hybrid dynamic model is conducted from three aspects.

1) Under the stationary random load, the simulated dynamic deformation of the overall drum driving system is checked by 3D plot.

Figure 9, which is drawn by MATLAB software using the simulation result under stationary random load, illustrates two instantaneous dynamic deformations of the overall drum driving system. The deformation in Fig. 8 has been magnified 350 times for clarity. It can be found that the deformation of ranging arm housing looks like a cantilever beam. Considering the three-directional cutting force applied on the drum (see Fig. 7), the deformation shown in Fig. 9 conforms to common sense. So Fig. 9 indeed contributes to validate the correctness of the modeling process.
2) Under the stationary random load, relative relation of the amplitude among horizontal vibration, vertical vibration and axial vibration at the similar points on experimental housing and simulation housing are compared.

Figure 10 illustrates the housing vibration under the stationary random load. Figure 10(a) is the experimental data (Fig. 8 point A), which indicates that axial vibration is the largest, while vertical vibration is the smallest. Figure 10(b) is the simulated vibration at a similar point on the range arm housing. The relative relation of the amplitude among three directions from simulation coincides with the experimental result. So Fig. 10 indeed contributes to validate the correctness of the modeling process.

3) During a load increasing process, the relationship between cutting power and housing acceleration are analyzed through both experimental result and simulation result.

Figure 11 illustrates the histories of both the cutting power and housing acceleration. Because the cutting motor is an asynchronous motor without control, which works according to the torque-speed mechanical characteristic curve, the motor speed is assumed to be constant. The experimental cutting power is obtained by measuring the current and voltage of the motor, since the torque and speed are hard to get. The simulation cutting power is obtained by multiply the torque and speed. Since Fig. 11(a) is an intercepted section of the whole experimental test, the cutting power and housing acceleration are not zero at 0 second. From both experimental result (Fig. 11(a)) and simulation result (Fig. 11(b)), it can be observed that Root Mean Square (RMS) of the housing acceleration increases in proportion with the cutting power. In order to clearly display the relationship between cutting power and housing acceleration, the cutting power is equally divided into 44 levels and the housing accelerations at each level is plotted in Fig. 12. Both experimental result (Fig. 12(a)) and simulation result (Fig. 12(b)) reveal the positive correlation between cutting power and housing acceleration. So Fig. 11 and Fig. 12 indeed contribute to validate the correctness of the modeling process.
Because of the cyclic variation of the number of teeth actually involved in cutting and the non-uniform coal/rock strength, the actual cutting load fluctuates about its mean value. Define the ratio of the maximum load to the mean load as load fluctuation coefficient (LFC). Fig. 13(a) illustrates the dynamic deformation at bearing bore BP16 of the ranging arm housing (the number of the bearing bores is presented in Fig. 3) under different LFC value. As shown in Fig. 13(b), the mean value of the dynamic deformation is independent from the LFC, while the maximum value of the dynamic deformation increases almost linearly with the LFC. Since the mean value of cutting force is 511.0 kN, the static stiffness and dynamic stiffness at BP16 are 406.8 kN/mm and 115.7 kN/mm, respectively. The dynamic stiffness is much lower than the static stiffness.

In order to reduce the dynamic deformation of ranging arm housing, the first order natural frequency is set as the objective function to be maximized, and the total weight is set as the constraint in order not to increase the weight. Optistruct module in Hypermesh commercial software is adopted to solve the topological optimization problem. Figure
14 is the pseudo-density nephogram given by Hypermesh, element density near 1 means the material is more important to increasing the natural frequency, while element density near 0 means less important. The pseudo-density nephogram gives advice on the better material distribution for a housing with higher natural frequency.

According to the topological optimization result, moving the material from less important area to more important area can improving the natural frequency of housing. The original housing is given in Fig. 15(a); the optimal housing is given in Fig. 15(b). Look at the cut view section of optimal housing, four corners are strengthened while four edges are thinned. In order to avoid increasing the vibration greatly at bearing bore for thinner steel plate, some material is left by trial and error, which looks like the stiffeners.

The first 6 natural frequencies were compared between original housing and optimal housing, and most of them (except the 5th mode) are increased by 6.60% to 44.25% for the optimal housing, as shown in Table 1.

### Table 1 Natural frequencies for two housings

<table>
<thead>
<tr>
<th>Mode No.</th>
<th>Natural frequency, Hz</th>
<th>Increase, %</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Original</td>
<td>Optimal</td>
</tr>
<tr>
<td>1</td>
<td>27.14</td>
<td>39.15</td>
</tr>
<tr>
<td>2</td>
<td>41.06</td>
<td>46.87</td>
</tr>
<tr>
<td>3</td>
<td>81.84</td>
<td>88.21</td>
</tr>
<tr>
<td>4</td>
<td>106.59</td>
<td>121.74</td>
</tr>
<tr>
<td>5</td>
<td>197.07</td>
<td>190.64</td>
</tr>
<tr>
<td>6</td>
<td>242.99</td>
<td>259.03</td>
</tr>
</tbody>
</table>

### 5. Simulation and analysis

This section examines the influence of housing topological optimization on the gear meshing state. As coal seam is inhomogenous with hard inclusions and rock intercalations, the drum load can be represented by the random load with specific frequency components, shown in Fig. 16(a). Under rated condition, the mean value of random load is 515.1 kN·m. When the shearer cuts from soft coal seam to hard coal seam or encounters the fault, the drum load will increase immediately which can be represented by a step load, shown in Fig. 16(b). The immediately increase occurred at 5s,
drum load during 0~5 s is rated load which is the same as Fig. 16(a), drum load during 5~10 s is twice the load of 0~5 s. The load fluctuation coefficient (LFC) of 0~5 s and 5~10 s is the same, both are 1.15.

5.1 Influence of housing topology on dynamic deformation

Figure 17 compares the dynamic deformation at bearing bore BP17 (the number of the bearing bores is presented in Fig. 3) of the original housing and optimal housing under the random load. The static amplitude of deformation, which is 1.88 mm for original housing, reduces to 1.15 mm for optimal housing (a reduction of 38.8%). The dynamic amplitude of deformation, which is 0.89 mm for original housing, reduces to 0.55 mm for optimal housing (a reduction of 38.2%). The housing deformation has been reduced significantly by housing topological optimization.

5.2 Influence of housing topology on equivalent mesh misalignment

The equivalent mesh misalignment is largest at the gear pair Z4-Z5 (the number of the gear pairs is illustrated in Fig. 2) than other gear pairs. The time domain of equivalent mesh misalignment at gear pair Z4-Z5 under the random load for original housing is shown in Fig. 18(a). The maximum equivalent mesh misalignment (MAX) is 19.478 μm, and the dynamic factor of equivalent mesh misalignment (DYN) is 2.340. The time domain of equivalent mesh misalignment at gear pair Z4-Z5 under the random load for optimal housing is shown in Fig. 18(b). The maximum equivalent mesh misalignment (MAX) is 16.616 μm, and the dynamic factor of equivalent mesh misalignment (DYN) is 2.251.

Figure 19(a) illustrates the time domain of equivalent mesh misalignment at Z4-Z5 under the step load for original
housing, during 3~5 s the curve is the same as the same period in Fig. 18(a), when the drum load suddenly doubled at 5 s, the equivalent mesh misalignment increases immediately and then reaches the steady state slowly. The MAX during 8~10 s is 39.059 μm, which is almost twice (2.005 times) the 19.478 μm as the drum load doubled. It can be concluded that the maximum equivalent mesh misalignment is directly proportional to the mean value of the drum load when the load fluctuation coefficient (LFC) of the drum load is fixed. However, the MAX is 45.540 μm during 5~7 s, and the impact factor (IMP, given by Eq. (18)) is 1.166. Figure 19(b) illustrates the time domain of equivalent mesh misalignment at Z4-Z5 under the step load for optimal housing, the IMP is 1.100 which is lower than the original housing. Please see Table 2.

\[
\text{IMP} = \frac{\text{MAX}_{5-10s}}{\text{MAX}_{8-10s}}
\]

(18)

Fig. 19 Equivalent mesh misalignment under step load for (a) Original housing, and (b) Optimal housing. The maximum equivalent mesh misalignment reduced 17.08% for optimal housing.

<table>
<thead>
<tr>
<th>Load</th>
<th>Statistic feature</th>
<th>Original housing</th>
<th>Optimal housing</th>
<th>Reduction %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Random</td>
<td>MAX</td>
<td>19.478 μm</td>
<td>16.616 μm</td>
<td>14.69</td>
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<tr>
<td></td>
<td>DYN</td>
<td>2.340</td>
<td>2.251</td>
<td>3.80</td>
</tr>
<tr>
<td>Step</td>
<td>MAX_{8-10s}</td>
<td>39.059 μm</td>
<td>34.322 μm</td>
<td>12.13</td>
</tr>
<tr>
<td></td>
<td>IMP</td>
<td>1.166</td>
<td>1.100</td>
<td>5.66</td>
</tr>
</tbody>
</table>

Table 2 Statistic feature of equivalent mesh misalignment

5.3 Influence of housing topology on dynamic meshing force

The dynamic meshing forces at the gear pair Z4-Z5 of different housings are also compared. The time domain of dynamic meshing force under the random load of the original housing are shown in Fig. 20(a). The maximum dynamic meshing force is 208.58 kN. Figure 20(b) illustrates the dynamic meshing force of the optimal housing, and the maximum value is almost the same (208.36 kN). The same phenomenon can be observed in Fig. 21, the time domain of dynamic meshing force under the step load. It can be concluded that the influence of housing topological optimization on the dynamic meshing force is not very significant. This maybe because the contact ratio and pressure angle are insensitive to the housing deformation, so the dynamic meshing force is almost the same for the optimal housing.

Fig. 20 Dynamic meshing force under random load for (a) Original housing, and (b) Optimal housing. The maximum value is almost the same for optimal housing.
6. Conclusion

In this study, the hybrid dynamic modeling of housing-transmission coupled system is proposed. The following conclusions were obtained:

(1) For the steady state, the maximum equivalent mesh misalignment is directly proportional to the mean value of the drum load (in this study, the load fluctuation coefficient (LFC) of the drum load is fixed).

(2) When a step load is applied to the drum, the equivalent mesh misalignment increases immediately and reaches the steady state slowly. The impact factor (IMP) larger than one can represent the level of the transient maximum equivalent mesh misalignment larger than the steady maximum equivalent mesh misalignment, and it is the overshoot of the drum driving system to the sudden increase of drum load.

(3) The ranging arm housing topological optimization can reduce the dynamic deformation of the housing and thus also can reduce the equivalent mesh misalignment. The maximum value of equivalent mesh misalignment is decreased by 14.69% under random load, and the IMP of equivalent mesh misalignment is decreased by 5.66%.

(4) The influence of housing topological optimization on the dynamic meshing force is not very significant.

Acknowledgement

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